Dedendum Optimization of Asymmetric Spur Gear

Prateek Kumar, Saroj Mandal and S.Senthilvelan

Abstract

Due to the demand in increased load carrying capacity with reduced weight, size and vibration, asymmetric gear design is getting significance for the engineering applications where unidirectional load requirement exists. Unlike symmetric gears, different base circle diameter of drive and coast side gear tooth profile necessitates optimization of asymmetric gear root circle diameter. In this work, finite element analysis is performed to optimize the dedendum for the chosen module of 3, 18 teeth, 20-34° asymmetric spur gears.

Keywords: Asymmetric gear, Finite element analysis, Optimization

1 Introduction

In a standard symmetric gear, pressure angle of both drive and coast side of gear tooth profile are same, hence the bending and contact strength will also be same for both the sides of a gear tooth. However when the gears are used to transmit power in one direction, one side of the gear tooth (drive side) is significantly loaded for longer periods and other side (coast side) is loaded only for shorter period. In spite of this functional difference, standard symmetric gears are in use due to the intricacy in manufacturing the asymmetry gear (different pressure angles at coast and drive side). Mass production of these asymmetric gears can be economically carried out by injection molding/powder metallurgy process, due to the single die requirement. In the recent years, significance of asymmetric gear design is on the rise due to invent of high strength polymeric materials and utilization of injection molded gears for load carrying applications. By the asymmetric teeth, designers can reduce the space required, level of vibration and noise for the fixed requirement of motion and power transmission. In the recent years, many works have been reported on the design of asymmetric gears [1-10]. Litvin *et al* [1] proposed asymmetric gear design with suitable cutter geometry. Finite element analysis was carried out on 25-25, 35-25 and 20-25 with ABAQUS and examined contact and bending stress distribution. Kapelevich [2] developed theory for independently defining gear parameters from any generating rack parameter. Kapelevich also performed vibration test and confirmed that gear with larger pressure angle exhibited reduced vibration level due to the low sliding ratio and low meshing stiffness. Litvin *et al.* [3] used finite element method to simulate the meshing of asymmetric face gear used in helicopter application. Influence of alignment errors and bearing contact shift is examined with finite element analysis. Three dimensional asymmetric gear models were used to examine bending and contact stresses. Yang [4] used

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double envelope concept to design the internal gear with asymmetric profile. A mathematical model is developed and used for contact analysis. Finite element analysis of internal asymmetric gear and internal asymmetric helical gear was carried out with the aid of Nastron tool. Senthilkumar *et al.* [5] developed non standard asymmetric rack cutters for asymmetric involute surfaces and trochoidal fillet surfaces. With the aid of finite element analysis rack cutter shift was optimized and computed maximum fillet stresses were compared with the AGMA and ISO codes.

Kapat *et al.* [6] developed a dynamic model with MATLAB to predict instantaneous dynamic loads of spur gears with symmetric and asymmetric teeth. Dynamic factor increases with increasing pressure angles on the drive side of asymmetric spur gears. For asymmetric teeth, increasing the addendum leads to a significant decrease in the dynamic factor. Costopoulos and Spitas [7] designed a gear with involute profile at drive side and straight line profile at the addendum of coast side. Conventional trochoidal fillet of gears were is replaced with circular fillet and loading is assumed to act at the tooth tip. The FEA results obtained over a wide range of tooth numbers indicate a 28% decrease of the maximum pinion fillet bending stresses in comparison to standard designs. Wang and Lim [8] developed a nonlinear, time-varying lumped parameter dynamic model of hypoid gear pair systems with asymmetric mesh stiffness. Effects of asymmetric mesh stiffness parameters such as mean mesh stiffness ratio, mesh stiffness variation and mesh stiffness phase angle on the dynamic mesh force response and tooth impact regions were examined. Pedersen [9] designed a cutting tool so the root shape optimization of the gear tooth can be achieved. With a high drive side pressure angle the bending stress improvements is in the order of 40% independent of the number of teeth on the gear, with a high coast side pressure angle the improvement is roughly half the size. Alipiev [10] proposed a new method for the geometric design of symmetric and asymmetric involute meshing in which the contact ratio of the gear drive is equal to its potential. Using the proposed method, the areas of the realized potential and the areas of the existence of involute gear drives are defined. A characteristic feature of this model is that the gears are modified simultaneously in two directions (radial and tangential), and the processing of the pinion and the gear is performed by different rack-cutters. In a summary, most of the asymmetric gear design focused cutter design for machining steel asymmetric gears. The asymmetric gear chosen for the analysis in this work has a pressure angle of 20^0 and 34^0 on the two profiles. Due to the different base circle diameter at drive and coast side, there is a need to design the optimum dedundem for the chosen asymmetric gear, and finite element analysis is carried out to accomplish this.

2 Asymmetric Gear Design

In the present investigation, spur gear having module of 3 mm is chosen. In order to avoid interference for the standard 20° pressure angle, 18 numbers of teeth are chosen. It was decided to use maximum possible pressure angle at other side to get maximum advance of asymmetric gear. However major limitation of increasing the pressure angle is the reduction of gear tooth thickness at addendum circle. As the pressure angle increases, tooth shape becomes more and more pointed; consequently the top land becomes smaller and ultimately results in pointed tip, this phenomenon is termed as ‘peaking’. The peaking limit sets a boundary to the maximum magnitude of pressure angle.

Gear standard procedures IS, recommended that the tip thickness should be greater than equal to 0.2 times the module for the hardened gears. To avoid peaking, maximum possible pressure angle of 34° was decided for the chosen module (3 mm) number of teeth (18) and pressure angle at one side (20°). Addendum of 1 mm module was selected similar to the standard full depth tooth system. As per the AGMA design recommendations, 0.38 times the module was selected (1.14 mm) for the fillet radius for the chosen asymmetric gear. In the conventional standard symmetric gears, the radial distance between pitch and tip circle diameter i.e addendum is taken as one module. Similarly dedendum, the radial distance between pitch circle diameter and root circle diameter is considered as 1.25 module. In the asymmetry gear, the base circle diameter at drive and coast side are different as the pressure angle at drive and coast side are different. If same 1.25 module is fixed for the asymmetric gear dedendum, then the involute portion will be reduced at higher side pressure angle which may weaken the gear. Hence in this work, an attempt is made to optimize the root circle diameter/dedendum for the chosen asymmetric gear.

It is to be noted, due to the different pressure at drive and coast side, the base circle diameters at drive and coast sides (50.74 and 44.76 mm) are not same. However root circle diameter of the gear must be same. Initially, average of these two base circle diameters (47.04 mm) was taken as common root circle of asymmetric gear as shown in figure1. In the present work, by keeping all other gear parameter as constant (module, pressure angle at drive and coast side, root fillet radius, addendum) evaluation of best root circle diameter is attempted. Root circle diameter was varied by varying the dedendum from 1.05m to 1.35m in step increments of 0.5m by keeping all other gear parameters as constant. The fillet radius was kept constant at (0.25 times the module) 0.75 mm for all of the designs.

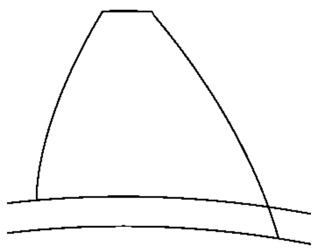

Figure 1a: Different base circle diameter of drive and coast side of asymmetric gear tooth

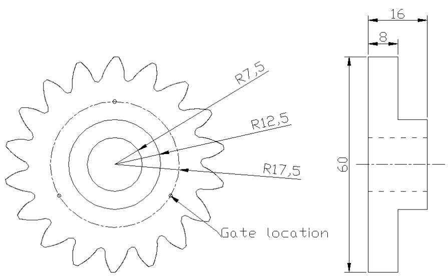

Figure 1b: Detailed of chosen asymmetric gear

3 Finite Element Model of Asymmetric Gear

It is necessary to maintain similar size and number of elements in the crucial fillet area for all the proposed design. Gear tooth was discretized in such a way that the fillet region remains unchanged for all of the possible designs. Eight noded, quadratic plane 82 element was considered for this analysis and plane stress with 0.5 thickness option was chosen. Thermoplastic material, polypropylene was chosen with E as 2 GPa and Poisson’s ratio as 0.3 and the material was assumed to be linear, elastic and isotropic. The accuracy of this mesh size is studied by checking the convergence of maximum stress along drive side fillet radius. During meshing, the small element type was chosen as quadrilaterals and mapped mesh was applied. The bottom area which is part of the gear body was meshed as free.

Proposed discretized models were subjected to constant load A uniform torque of 1.5 Nm was applied to the different gears.

$$T=F_t * r \tag{1}$$

Where

r, pitch circle radius, 0.027 m

T, applied torque

$F_{t,}$ Tangential force applied on the gear.

Now contact width of the gear at the pitch circle diameter = b

$$b=\sqrt{\frac{8F}{\pi L}*\frac{(1-v1^2)/E1+(1-v2^2)/E2}{\frac{1}{D1}+\frac{1}{D2}}} \tag{2}$$

Where

$E1$,$E2$, Modulus of Elasticity of the mating gears, 2 GPa for polypropylene

$v1$,$v2$,Poisson's ratio of polypropylene gear, 0.3

$D1$,$D2$= Equivalent diameters of the polypropylene gears.

$$D1=D2=d\sin\alpha \tag{3}$$

Here d= pitch circle diameter

α = Drive side pressure angle of the gear tooth, 20^0

At the pitch point, width for the symmetric and asymmetric is found to be 0.275mm and 0.373mm respectively. Now width of the tooth w=0.008 m

Area where pressure, p is applied is

$$a=w*b \tag{4}$$

Pressure, p=F_t/a, this pressure was applied over the computed tooth width to evaluating the gear tooth stress

4 Results and Discussions

Figure 2a and 2b shows the developed discretized model of asymmetric gear and developed stress pattern along a triangular area at the fillet of asymmetric gear when load is applied at drive side. Figure 3 shows the induced Von Mises stresses along the fillet for the various considered gear design (dedendum ranging from 1.05 m to 1.35 m). Figure 3 shows that dedendum of 1.1 m exhibited less stress among all the other designs. As the dedendum increases from 1.05 m, stress decreases first then after reaching 1.15 it increases further.

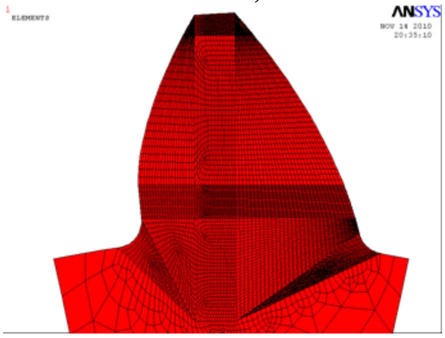

Figure 2a: Finite element model of single gear tooth

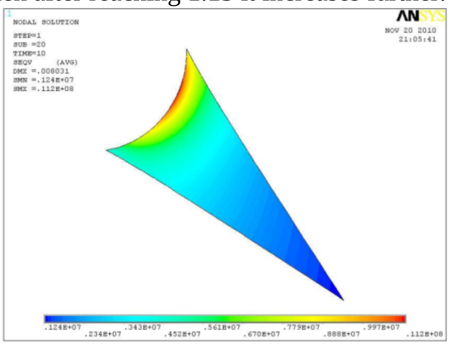

Figure 2b: Identification of maximum stress along the fillet Stress at 1.05m dedendum

gear

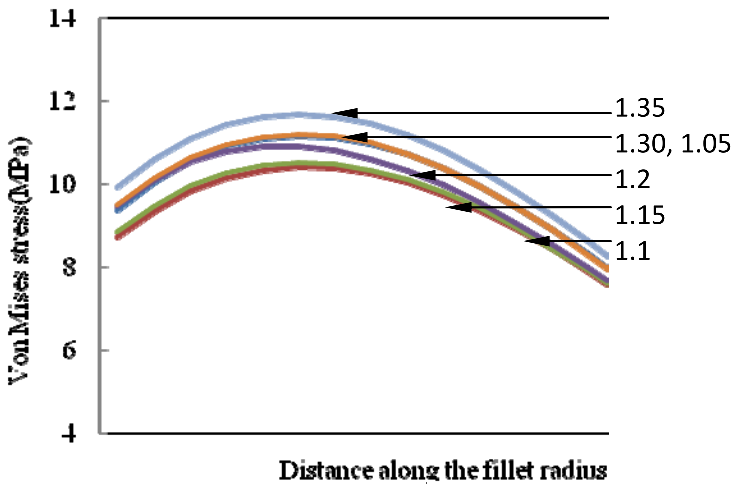

Figure 3: Induced stress at drive side fillet profile

This behavior is explained by considering minimum dedendum (1.05m) and maximum dedendum (1.35m). Figure 4 (a-b) shows the schematic of considered gear design. Two factors contribute to the generation of stress at the fillet, moment distance and the cross sectional area. The 1.05 design has smaller moment distance, so the bending moment is less, but the cross sectional area is also considerably less. So the net effect is a large stress at the fillet. The 1.35 design has a larger moment distance, but the cross sectional area at the fillet is more. Here the bending moment factor dominates and the stress is again large. Hence the optimum design must lie in between these two extremes, which is found to be 1.1m. The stress profile pattern at the fillet edge of the coast side is similar to the drive side, but the magnitude was higher than that at drive side (Fig. 5).This behaviour is due to the larger neural axis distance as shown in the schematic figure (Fig. 6).

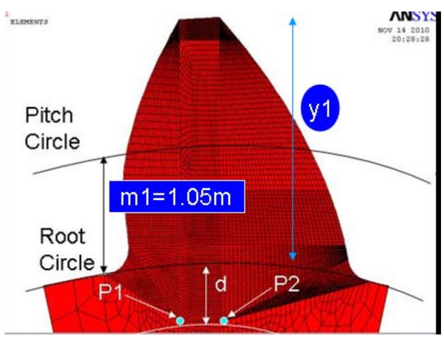

Figure 4a Schematic of 1.05m dedendum gear

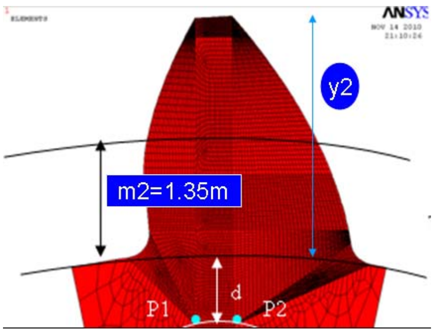

Figure 4b: Schematic of.35m dedendum gear

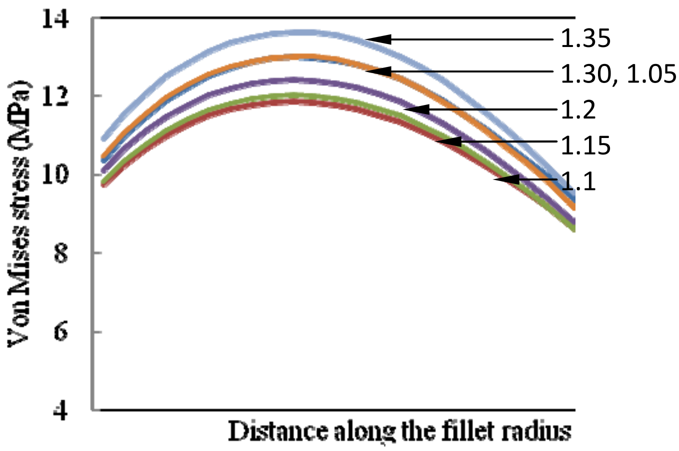

Figure 5: Coast side fillet profile for the different designs

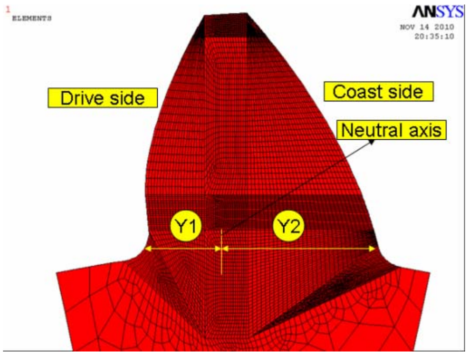

Figure 6: Fillet distance from the region of load application

To quantify the advantage of asymmetric gear, symmetric gear was also modeled with 20^0 pressure angle at both the side on each side. For the purpose of comparison same amount of dedendum 1.1 m was kept for the symmetric gear.

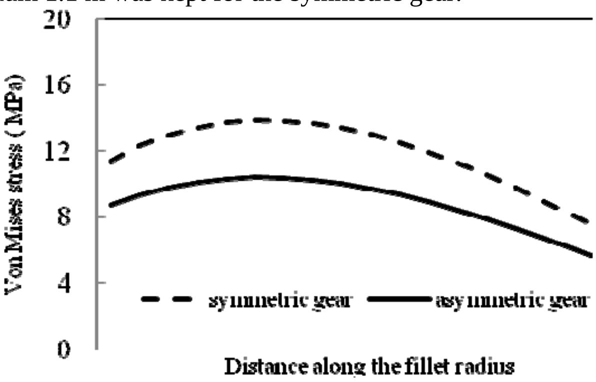

Figure 7: Drive side stress: Symmetric and asymmetric

From Figure 7 and 8, it is revealed that there is about 40 % stress increase in the symmetric gear compared to that of asymmetric gear. This behavior is due to the smaller root section(XY) when compared to that of asymmetric gear(X'Y'). Besides though there is constant fillet radius, there is a gradual area change at point Y' than that Y.

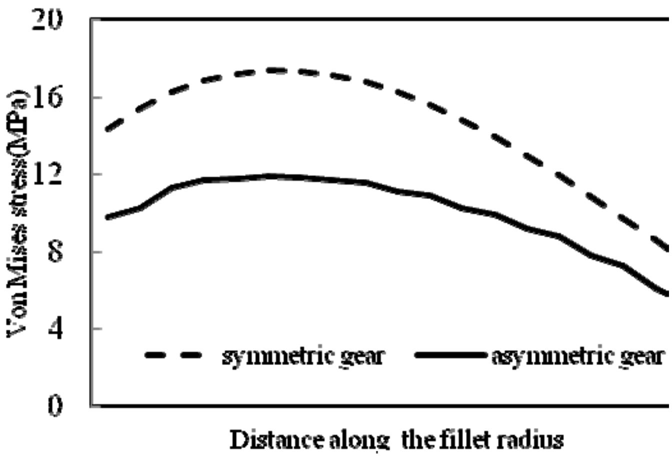

Figure 8: Coast side stress- symmetric and asymmetric

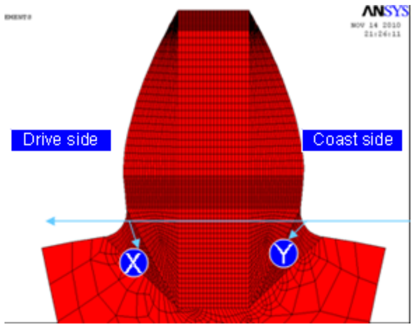

Figure 9a: Schematic of symmetric gear root section

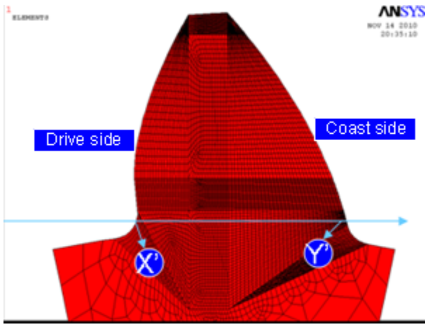

Figure 9b: Schematic of asymmetric gear root section

5 Conclusions

Finite element analysis of asymmetric gear was carried out to design the optimum dedendum. For the chosen 20-34 asymmetric gear pair, 1.1 module is identified as optimum dedendum. By fixing all other gear parameters, 20-20 gear tooth was compared with 20-34 gear tooth to quantify the asymmetric gear advantage.

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